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# Experimental Investigation of Particle Emissions from a Dieseline Fuelled Compression Ignition Engine

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## ABSTRACT

Achieving low-smoke and low-$NO_x$ premixed compression ignition (PCI) combustion at a wide engine operating load range has been a challenge; especially in multi-cylinder engines running at higher loads for which less data is available in the literature. More specifically, it is of interest to characterise particle emissions under these conditions and identify their possible reduction benefit in different size classes compared to conventional diesel combustion. Mixing diesel with gasoline (Dieseline) as an incentive to reduce fuel reactivity (cetane-number) and consequently improve premixing is believed to be useful for PCI. In this study, the feasibility and benefits of using low cetane-number (<30) and wide boiling range G75-Dieseline (75% gasoline in diesel based on volume) in a production light-duty 4-cylinder CI engine are investigated at medium-high loads of 6, 12 and 17.3 bar BMEP. It was found that G75 combustion resulted in lower particle emissions (both number and mass), by up to 99.5%, while maintaining the same range of efficiency and $NO_x$ compared to diesel combustion. Bimodal particle size distributions were observed for both G75 and diesel while concentrations of G75 particles were much lower across the entire diameter range. For G75, increase of fuel injection pressure decreased particle number concentration (especially in nucleation mode) while particle mass was less affected. At medium loads, because of longer ignition-dwell of G75 compared to diesel, variations of combustion and emission characteristics were more sensitive to injection timing. At high loads, mixing-controlled combustion phase was observed for G75 and highlighted the importance of investigating advanced intake pressure boosting systems and interactions between fuel spray and piston.

**Keywords:** PCI; Dieseline; Low cetane, Injection strategy; Particulate matter; $NO_x$

# 1 INTRODUCTION

Using premixed compression ignition (PCI) combustion techniques can effectively decrease engine-out soot and oxides of nitrogen ($NO_x$) emissions compared to conventional diesel CI [1-7]. This is normally achieved by using longer fuel ignition-delay (enhanced local premixing) and higher exhaust gas recirculation (lower intake $O_2$ concentration and local combustion temperature) [7]. Generally, engine operating range is a challenge to overcome when using PCI techniques as they are normally limited to low, not very low though, and medium loads. At higher loads, the global and local equivalence ratios are closer to stoichiometric strength and auto-ignition of multiple points happens more rapidly leading to high peak heat release rates, pressure rise rates, $NO_x$ and possibly particle emissions [1, 8]. Low auto-ignition tendency (low cetane-number (CN) or long ignition-delay (ID) or low reactivity) and high volatility of the fuel can help achieve the objectives of PCI type combustion in a wider load range [1].

Results from the recent research in the area of PCI suggest usage of a gasoline-like fuel with a research octane number (RON) between 70 and 85 [1, 9]. Among the readily available options, mixing of diesel with gasoline (named as Dieseline) seems to be a very promising choice to increase the reactivity of high RON conventional gasoline fuel. Using Dieseline has been proven to enhance the PCI combustion at low-medium loads and is being considered as one of the fuel candidates for the future use (e.g. [5, 6, 10-16]). Various gasoline volumetric blend-ratios in the Dieseline fuel has been studied for PCI at low-medium loads, e.g. between 0% and 75% (G0-Dieseline and G75-Dieseline) [5-7, 12, 13, 17]. These studies showed that smoke and $NO_x$ emissions were reduced, by up to 99% compared to neat diesel combustion, especially for the case of G75-Dieseline (G75) [7] that has an estimated RON of approximately 75 and consequently long ID. In addition to these studies at low-medium loads, possible application of the Dieseline PCI technique in multi-cylinder engines requires investigating combustion performance and emissions characteristics at higher loads. It is particularly important to evaluate the reduction of particles compared to diesel combustion and its impact on engine efficiency and $NO_x$ emissions.

A few studies are available on the particle emissions from CI engines using fuels that contain only small to moderate (not high) gasoline blend-ratios in Dieseline at medium-high engine loads (e.g. [18-21]). Wei *et al.* [18] studied a maximum gasoline blend ratio of 30% at a maximum brake mean effective pressure (BMEP) of 11.3 bar without any exhaust gas recirculation (EGR) in a light/medium-duty CI engine. They concluded that total concentration of particles with smaller diameters (mainly in the nucleation mode) was generally higher than the case of diesel combustion. Belgiorno *et al.* [19] investigated the mixtures of diesel, gasoline and ethanol (with a maximum gasoline+ethanol ratio of 44%) in a single-cylinder CI engine at a maximum load of 13 bar BMEP. They reported that nucleation mode particles were higher than diesel in lower loads while accumulation mode particles were higher at higher loads. Benajes *et al.* [20, 21] investigated dual-mode/dual-fuel (combining reactivity controlled CI (RCCI [22]) and diffusive combustion) in a single-cylinder medium-duty CI engine at a maximum load of 22 and 23 bar indicated MEP for which gasoline ratio in Dieseline was 34% and 32%. Total particle number results were higher than conventional diesel combustion. Although gasoline blend-ratio was different for different loads, they concluded that nucleation mode particles and accumulation mode particles were higher in lower and higher loads, respectively.

The results from these recent studies motivated the current research into particle emissions from combustion of Dieseline with higher blend-ratios of gasoline at medium-high engine loads especially in multi-cylinder CI engines. It is of interest to characterise particle emissions and identify their possible reduction in all size classes.

In the current paper, PCI combustion of G75-Dieseline is evaluated in terms of efficiency and emissions. Different fuel injection pressures and timings with various EGR options are investigated in a production light-duty multi-cylinder diesel engine at loads of 6, 12 and 17.3 bar BMEP. The experimental setup is described in the next section and is followed by the comparison and discussion of G75 combustion and diesel combustion results. A summary and conclusions of this paper are provided at the end.

## 2 EXPERIMENTAL SETUP

Experiments were carried out on a production 2.2 L, 4-cylinder in-line compression ignition (CI) engine. It is equipped with a variable-nozzle-turbine (VNT) turbocharger and a common rail direct-injection (DI) system. Major engine specifications are provided in Table 1 and a schematic of the engine test cell is shown in Figure 1; further details are described in [7]. In this paper, hot-EGR means that the EGR cooler and air intercooler are either completely or partially deactivated by means of the valves illustrated in Figure 1. Cold-EGR term is used when full cooling intensity of the EGR cooler and the air intercooler are used.

**Table 1** Engine specifications

| | |
|---|---|
| **Bore (mm)** | 86.0 |
| **Stroke (mm)** | 94.6 |
| **Connecting Rod Length (mm)** | 155.0 |
| **Displacement ($cm^3$)** | 2198 |
| **Compression Ratio** | 15.5:1 |
| **Injection System** | DI Common Rail |
| **Injectors** | Solenoid, 7 Holes (0.15mm diameter) |

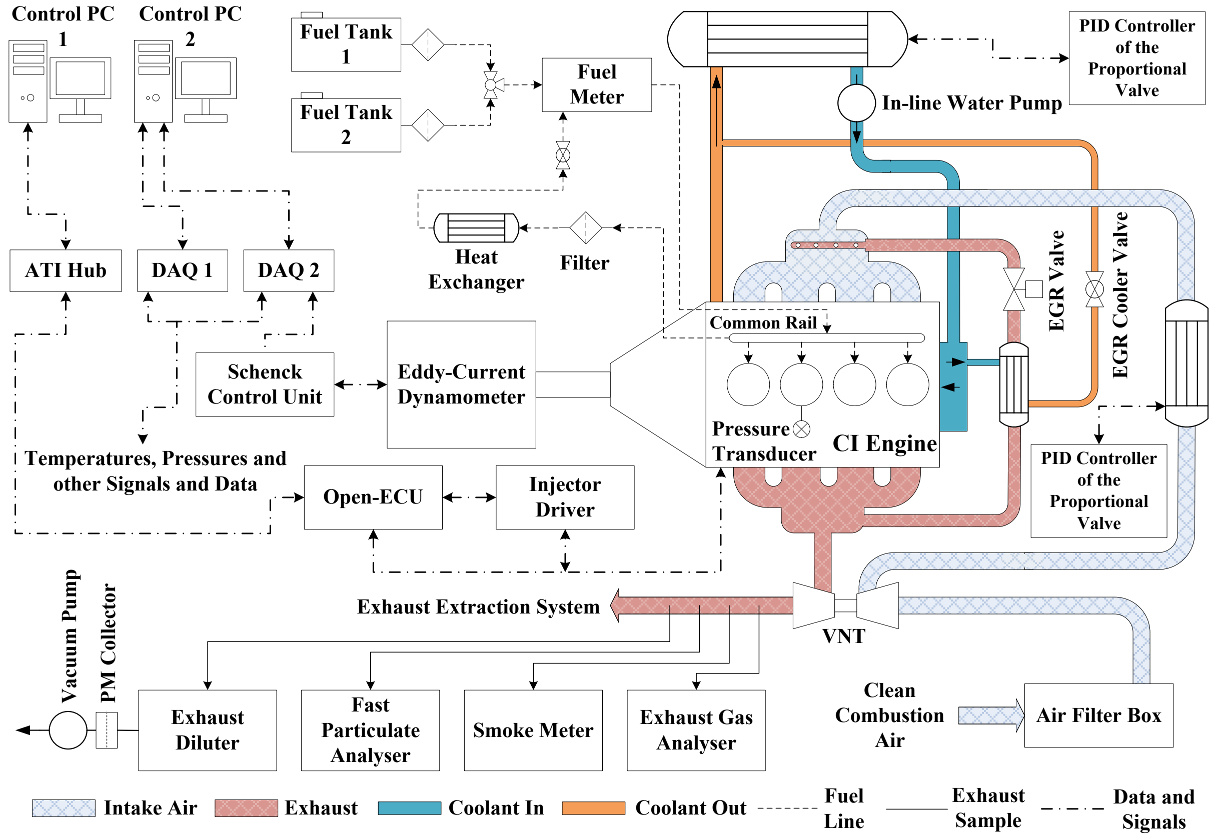



**Figure 1** Schematic of the engine test cell

A Pi-Innovo M250 open engine control unit (Open-ECU) is utilised in this test cell and allows flexible control over the engine operating settings, e.g. injection events, EGR valve, VNT actuator, etc. Fuel injection-timing (IT) is obtained from the Open-ECU which indicates the start of the energising.

The clean combustion air shown in Figure 1 is from the lab air supply system and its temperature was fixed at 22±1 ºC. Temperature of the engine coolant out was controlled to be 90±1 ºC. K-type thermocouples were used to measure temperature at different locations and at each test point, data were averaged for 180 s.

In-cylinder pressure of the engine was measured with a calibrated Kistler 6058A non-cooled piezo star pressure transducer (linearity ≤ ±0.05% of the full scale output) equipped with a

Kistler 6544Q192 glow-plug adaptor and a Kistler 5011B10Y50 charge amplifier (linearity ≤ ±0.05%). An AMI-Elektronik/Art.No:41500043-00360 shaft encoder was used and in-cylinder pressure data were logged at each crank angle degree (CAD) for 200 engine operating cycles. These data were used to calculate some of the combustion related parameters, e.g. apparent net heat release rate [7, 23].

Start of combustion (SOC) is the CAD at which heat release rate curve passes the zero level from a negative value to a positive value after the start of fuel injection. Ignition-delay (ID) is the CAD duration from the start of injection-timing to the SOC. Combustion-phasing or AHR-50 is defined as the CAD at which 50% of the accumulative heat release is achieved. Ignition-dwell is defined as the CAD duration from the end of injection (EOI) to the SOC.

Fuel consumption was measured and averaged over 180 s using a frequently calibrated AVL 733s dynamic fuel meter (error between 0.12% and 0.2%) equipped with an AVL 752-60 fuel cooler. A Horiba MEXA-7100-DEGR exhaust gas analyser (measurement linearity ≤ ±1% of the full scale output) was used to measure gaseous emissions and data were averaged over 180 s. $CO_2$ concentrations in the intake manifold (after the entry of the EGR tube) and exhaust were used to calculate the EGR percentage. Smoke emissions were measured using an AVL smoke meter (model 415S). In terms of repeatability, the standard deviation (SD) of the measurements by the smoke meter is < ±0.005 FSN + 3% of the measured value.

A calibrated fast particulate analyser from Cambustion (DMS500 MKII) working based on the differential mobility spectroscopy (DMS) principle was used to measure particle emission (from sizes around 5 nm to 1000 nm) from the engine exhaust. It separates different sizes of particles based on their charge and aerodynamic drag while migrating in an electric field [24]. Particle size distribution data are averaged over 60 s for a single measurement while each data point presented in this paper is the average of multiple measurements. Total particle number concentrations are derived from integrating the data while nucleation and accumulation mode concentrations are derived from log-normal curve fittings. A software package provided by Cambustion utilising a Bayesian statistical algorithm [25] was used to separate the two aforementioned modes based on the concentration, mean size and width (geometric standard deviation) of the distribution [24]. In this way, total mass can be calculated more accurately, as explained in [26], since the characteristics (e.g. effective density and physical geometry) of particles of each mode are different.

The experimental data presented in this paper were collected under the steady-state engine operating conditions and are averages of multiple measurements (at least three) while considering the standard deviation.

Specification EN 590 normal European ultra-low sulphur diesel (ULSD) and G75-Dieseline (G75) fuels were used in this study. G75 is a blend of 75% (based on volume) neat normal European RON95 unleaded gasoline (ULG95) (specification EN 228) in the ULSD. To avoid possible failure of the high pressure injection system, ULG95 was enriched with 300 volumetric parts per million of the Paradyne R655 fuel lubricity improver. Major available properties of the utilised fuels are provided in Table 2 [7]. The boiling curves of the utilised fuels have been presented elsewhere [7].

**Table 2** Properties of the utilised fuels [7]

| | **Diesel** | **G75-Dieseline** | **Gasoline** |
|---|---|---|---|
| **Density at 15 °C (kg/m$^3$)** | 835.1 | 768.2 | 742.8 |
| **RON (-)** | - | ~75* | 95.4 |
| **MON (-)** | - | - | 86.6 |
| **CN (-)** | 52.6 | - | - |
| **Derived CN (-) ×** | - | 30.0 | - |
| **Cetane Index (-)** | - | 27.2 | - |
| **Net Calorific Value (MJ/kg)** | 42.72 | 42.52 | 42.34 |
| **Total Paraffins (v/v %)** | - | 57.8 | 47.1 |
| **Olefins (v/v %)** | - | 7.2 | 7.9 |
| **Naphthenes (v/v %)** | - | - | 6.3 |
| **Aromatics (v/v %)** | - | 31.7 | 26 |

* this value is estimated based on the equation provided in [27] considering Cetane Index

× DCN measurement is certified for 33 < DCN < 64; at DCN= 34, the reproducibility is ±2.21

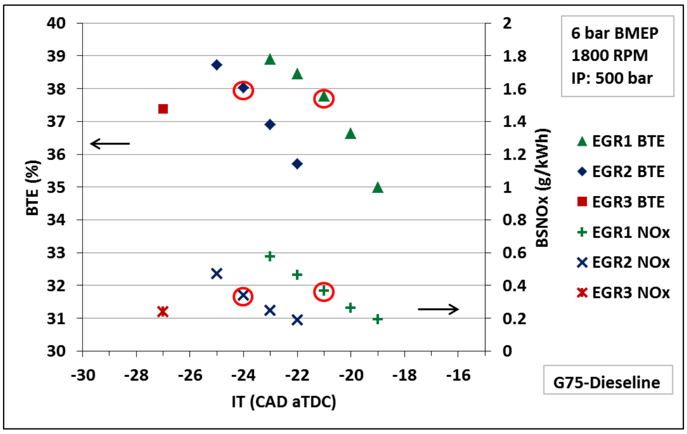
- means the value is not available

In this paper, G75 and diesel combustion are investigated at the engine loads of 6, 12 and 17.3 bar BMEP and engine speed of 1800 revolutions per minute (RPM) using single-injection. 17.3 bar BMEP is the maximum possible load that can be tested in the current engine test cell. Screening experiments at loads more than 6 bar BMEP showed that using low IPs (150, 250 and 350 bar) and/or hot-EGR strategy were not suitable for obtaining low smoke and $NO_x$ emissions from the G75 PCI combustion, as opposed to observations at lower loads [7]. Therefore, higher IPs and cold-EGR strategy were used for all of the tests conducted at these loads.

# 3 RESULTS AND DISCUSSION

## 3.1. Results at 6 bar BMEP

Figure 2 illustrates brake thermal efficiency (BTE) and brake specific (BS) $NO_x$ results for G75 fuel using three IPs, three EGR valve settings (referring to the opening level of the EGR valve) and various ITs. EGR1 refers to 48.5%, EGR 2 refers to 50.5% and EGR3 refers to 52.5% valve opening of the full opening position of the EGR valve. The actuation level of the turbocharger vanes was kept constant for all of these experiments. The reason for selecting this strategy was the fact that maintaining the EGR rate at a fixed level when the IT was being modified was challenging since there is an interaction between the EGR valve and the turbocharger in the multi-cylinder production engine. Therefore, it was decided to maintain valve positions at some defined settings concluded from several screening experiments and this was beneficial for maintaining specific air-fuel ratio ($\lambda$) at the desirable levels. With these settings, $\lambda$ was between 1.2 and 1.4 while $\lambda$ of EGR1>EGR2>EGR3 at the same combustion-phasing. EGR percentage was between 30% to 34% and engine intake pressure was between 1.20 to 1.28 bar absolute. At a fixed EGR valve opening level while using different ITs, EGR percentage variation was generally less than 1% (in absolute value). These strategies were proved to be effective for showing a clear variation trend of engine performance and emission characteristics when using different EGR rates, IPs and ITs. The range of ITs presented in Figure 2 was chosen based on two selected constraints: maximum pressure rise rate (MPRR)<12 bar/CAD and BTE>33%.

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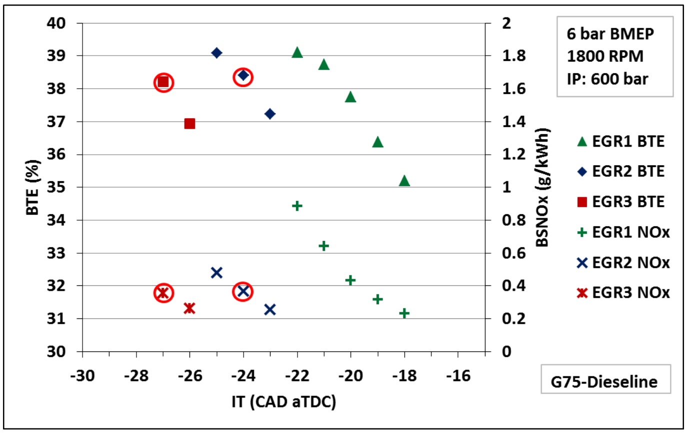


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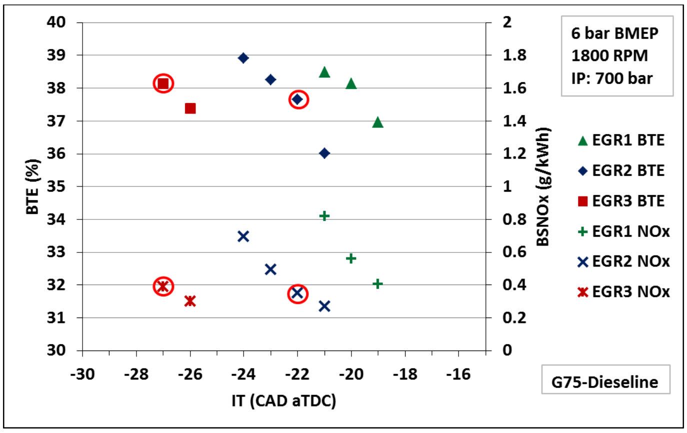


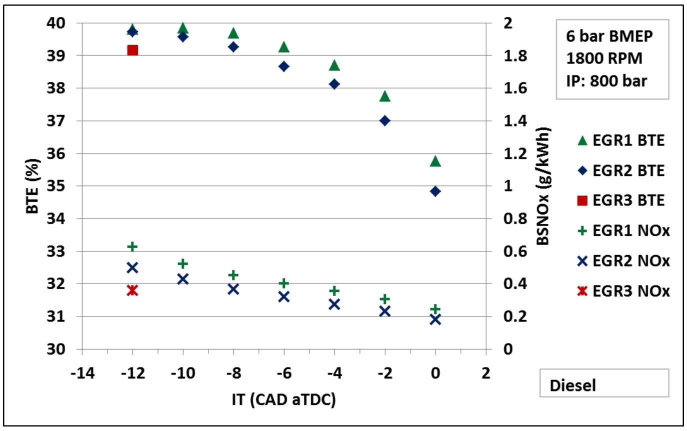

(c)

**Figure 2** BTE and $BSNO_x$ for G75 with different EGR valve positions and injection-timings with injection pressure of: (a) 500 bar, (b) 600 bar and (c) 700 bar at 6 bar BMEP; red circles in the figure mark the highest achieved BTE while $BSNO_x$ was below 0.4 g/kWh

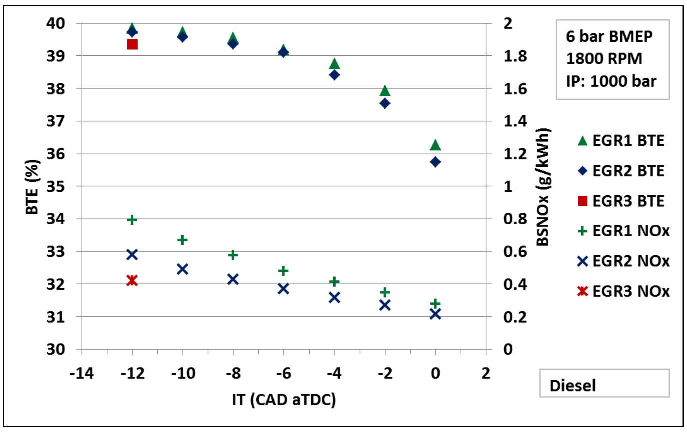
Based on Figure 2, at each IP, BTE and $BSNO_x$ were functions of IT and EGR valve position. Advancing IT increased BTE and $BSNO_x$ emissions. This is because combustion-phasing was advanced as earlier IT was used and therefore more expansion work was drawn from the combusting mixture while in-cylinder temperature at the point of combustion was higher. More opening of the EGR valve helped to achieve more advanced ITs without surpassing the 12 bar/CAD constraint for MPRR due to prolonged ignition-delay. With the same IT, increasing EGR valve opening and consequently reducing $\lambda$ resulted in lower $BSNO_x$ and BTE as expected. This is mainly due to the fact that lower intake $O_2$ concentration decreases the rate of $NO_x$ production [28] and reduces combustion efficiency. In general, with the same IT and EGR settings, increasing IP resulted in higher values of $BSNO_x$ and BTE as the AHR-50 was being advanced (approached towards the top dead centre (TDC)). A higher IP is associated with shorter injection durations (at a fixed engine load) and can help prepare fuel-air packets which are more readily combustible in a shorter period of time.

It can be concluded that there was a trade-off between increasing BTE and reducing $NO_x$ at 6 bar BMEP for G75 PCI combustion. This trade-off was not evident in lower engine operating loads [7]. Considering the $BSNO_x$ target value defined arbitrarily to be less than 0.4 g/kWh, within the studied test points when using IP of 600 bar, IT of -24 CAD after TDC (aTDC) and EGR rate of 32.74%, a maximum BTE of 38.41% was achieved and its corresponding $BSNO_x$ value was 0.36 g/kWh.

Figure 3 shows BTE and $BSNO_x$ results for diesel with the same EGR valve strategy used for G75 but with different ITs and IPs. Using the same IPs which were used for G75 resulted in high smoke emissions at the same AHR-50 while $BSNO_x$ emissions were around 0.4 g/kWh. Therefore, higher range of IPs were utilised which are normally used in diesel engines for this speed and load [5, 13]. With respect to G75, ITs had to be retarded to avoid high MPRR and intense combustion before the TDC due to the shorter ignition-delay of the diesel fuel.




(a)

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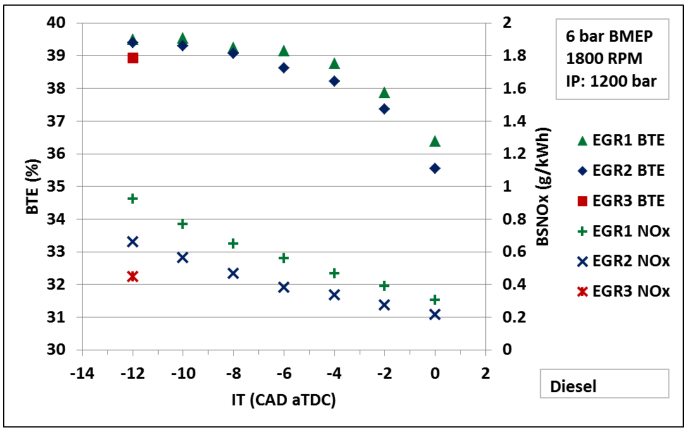



(c)

**Figure 3** BTE and $BSNO_x$ for diesel with different EGR valve positions and injection-timings with injection pressure of: (a) 800 bar, (b) 1000 bar and (c) 1200 bar at 6 bar BMEP

Considering the $BSNO_x$ target of <0.4 g/kWh, BTE levels for diesel were in the same range as G75 results (between around 38% to 39%) while smoke emissions were considerably higher (illustrated in Figure 4). Similar to the case of G75, using more advanced ITs resulted in higher BTE and $BSNO_x$ values. Increasing the IP of diesel fuel, from 800 bar to 1200 bar, resulted in an earlier AHR-50 and consequently higher $BSNO_x$ when using a fixed IT and EGR valve opening. For diesel, increasing the IP resulted in higher MPRR which can be linked to the combustion induced noise. These results indicate the effectiveness of AHR-50, $\lambda$ and IP for controlling combustion and emissions characteristics.

Smoke emission results for both G75 and diesel are illustrated in Figure 4. G75 combustion resulted in very low smoke values (mostly below 0.04 FSN) using the studied IPs, ITs and EGR valve openings. Diesel fuel combustion resulted in higher smoke emissions mainly due to its shorter ID and consequently fuel-air mixing time. For diesel, utilising higher IP combined with retarded IT helped to reduce smoke, however smoke emissions never reached below 0.5 FSN. These reductions in FSN were obtained with the lowest studied EGR valve opening (which normally resulted in higher $BSNO_x$) and retarded IT (which normally resulted in lower BTE). Thus, in the case of diesel combustion at 6 bar BMEP, simultaneous reduction of smoke and $NO_x$ had a drawback of reduced BTE (similar to the observations at 1.4 and 3 bar BMEP [7]).

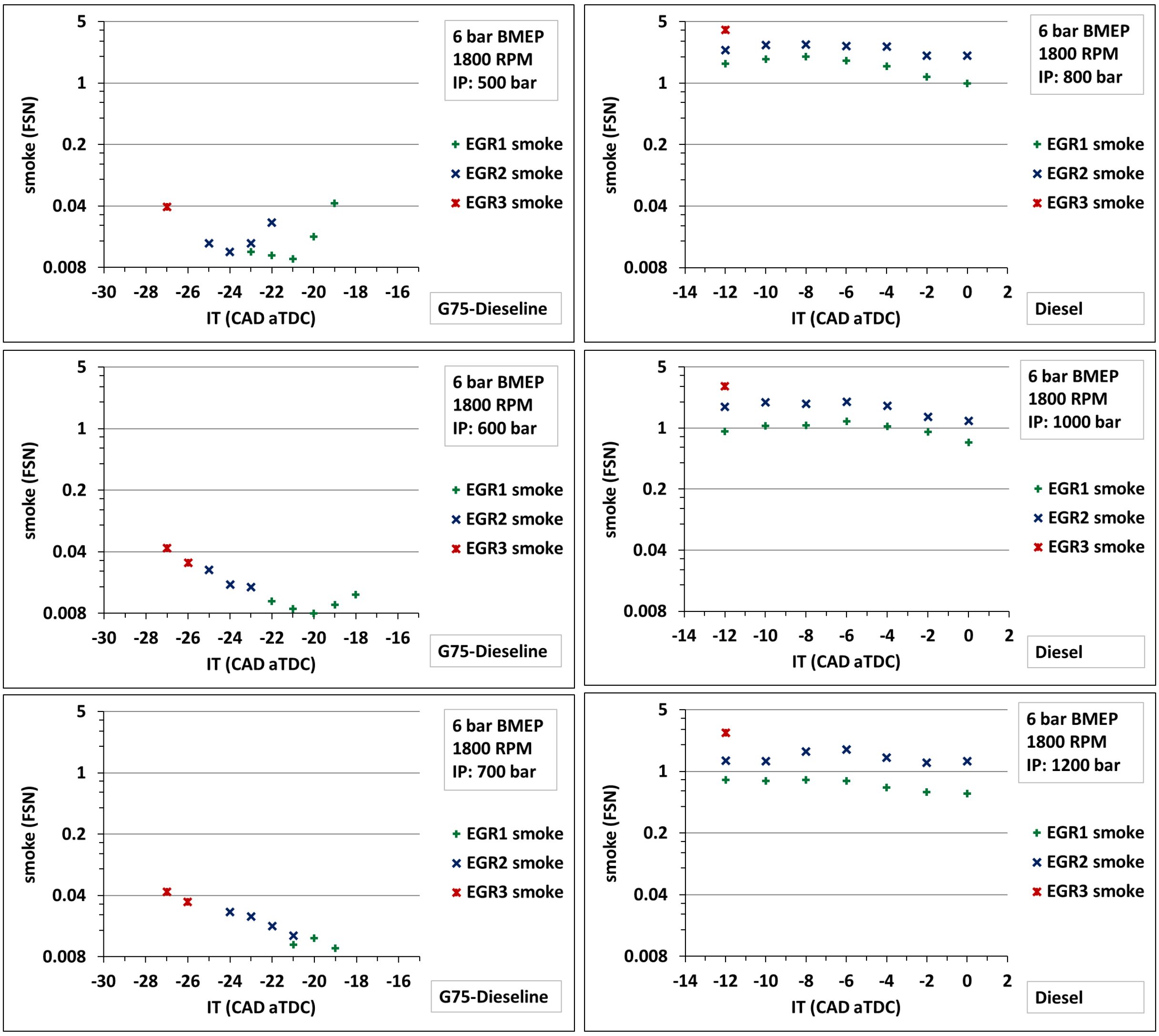



**Figure 4** Smoke emissions in terms of filter smoke number (FSN) for G75, left column, and diesel, right column, at different injection pressures, injection-timings and EGR valve positions at 6 bar BMEP (it should be noted that this graph is a semi-logarithmic plot)

In order to compare the mass of particle emissions from combustion of G75 and diesel fuels, brake specific accumulation mode particle mass (Acc. PM) emissions are illustrated in Figure 5. Agglomerates are believed to be the main contributor to the mass of total emitted particles from an engine [29]. The variation trend of Acc. PM was similar to the variation trend of smoke for both of the fuels. For G75 fuel, most of the studied engine operating conditions at 6 bar BMEP resulted in Acc. PM of less than 0.001 g/kWh while for diesel they were between 0.023

to around 1 g/kWh. Therefore, this figure confirms that PM emissions from G75 combustion were less than diesel by orders of magnitude which is obviously beneficial even when an exhaust particulate filter is used.

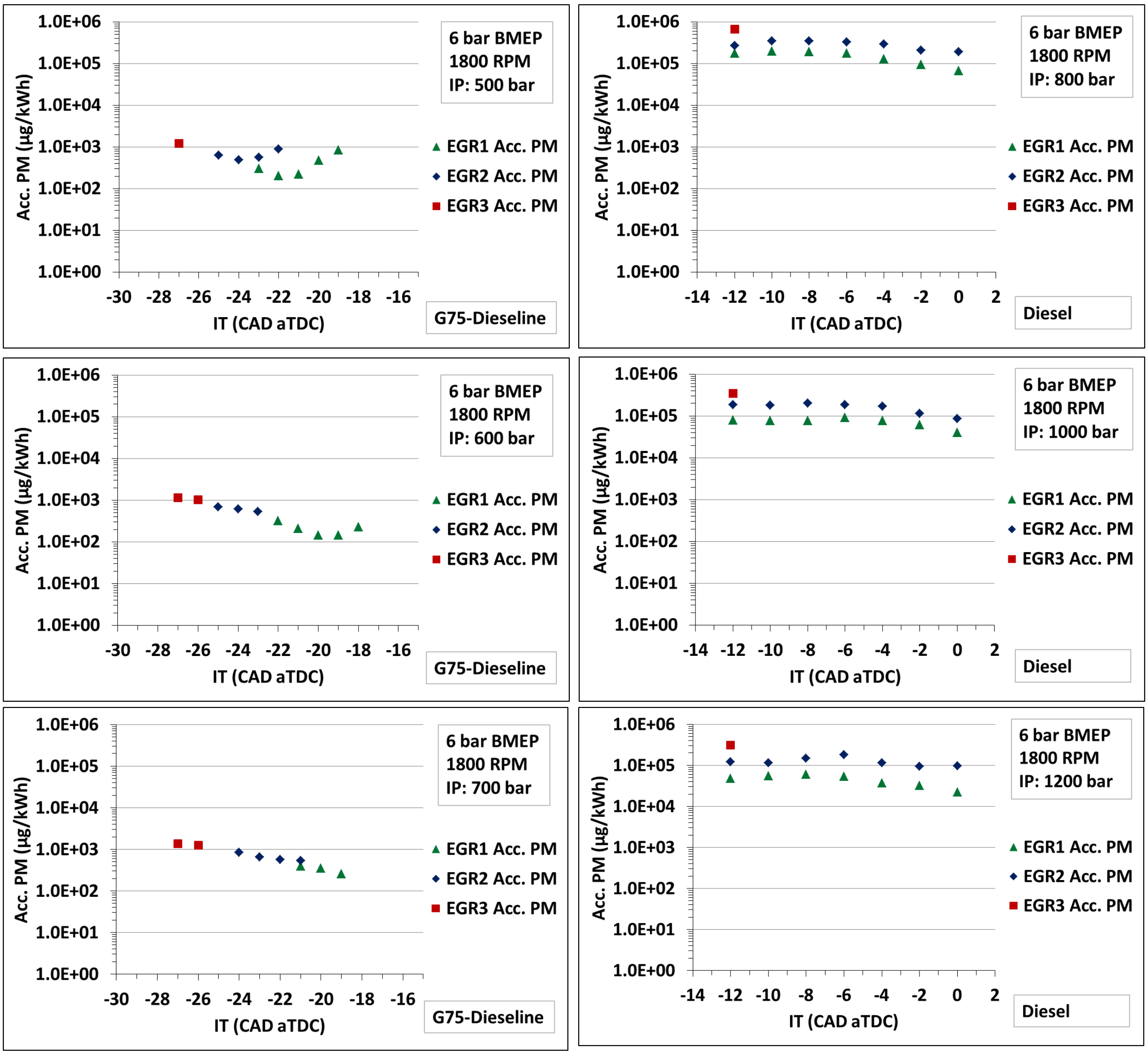



**Figure 5** Brake specific accumulation mode particle mass (Acc. PM) for G75, left column, and diesel, right column, at different injection pressures, injection-timings and EGR valve positions at 6 bar BMEP (it should be noted that this graph is a semi-logarithmic plot)

Brake specific total particle number (TPN) emissions are illustrated in Figure 6. These results show the number of detected particles in the diameter range of 5 to 1000 nm in both nucleation and accumulation modes. TPN emissions from G75 combustion were considerably lower than diesel. Similar to the trend for PM, for diesel at a fixed IT, more EGR valve opening resulted in higher TPN. This can be due to lower oxygen availability which results in more number of locally rich fuel-air packets and weaker soot oxidation. In general, using higher IP for both of the fuels resulted in lower TPN, probably due to better fuel and air mixing process. However, this trend was not significant for particle mass emissions from G75 combustion. This highlighted the necessity of investigating the size of these particles.

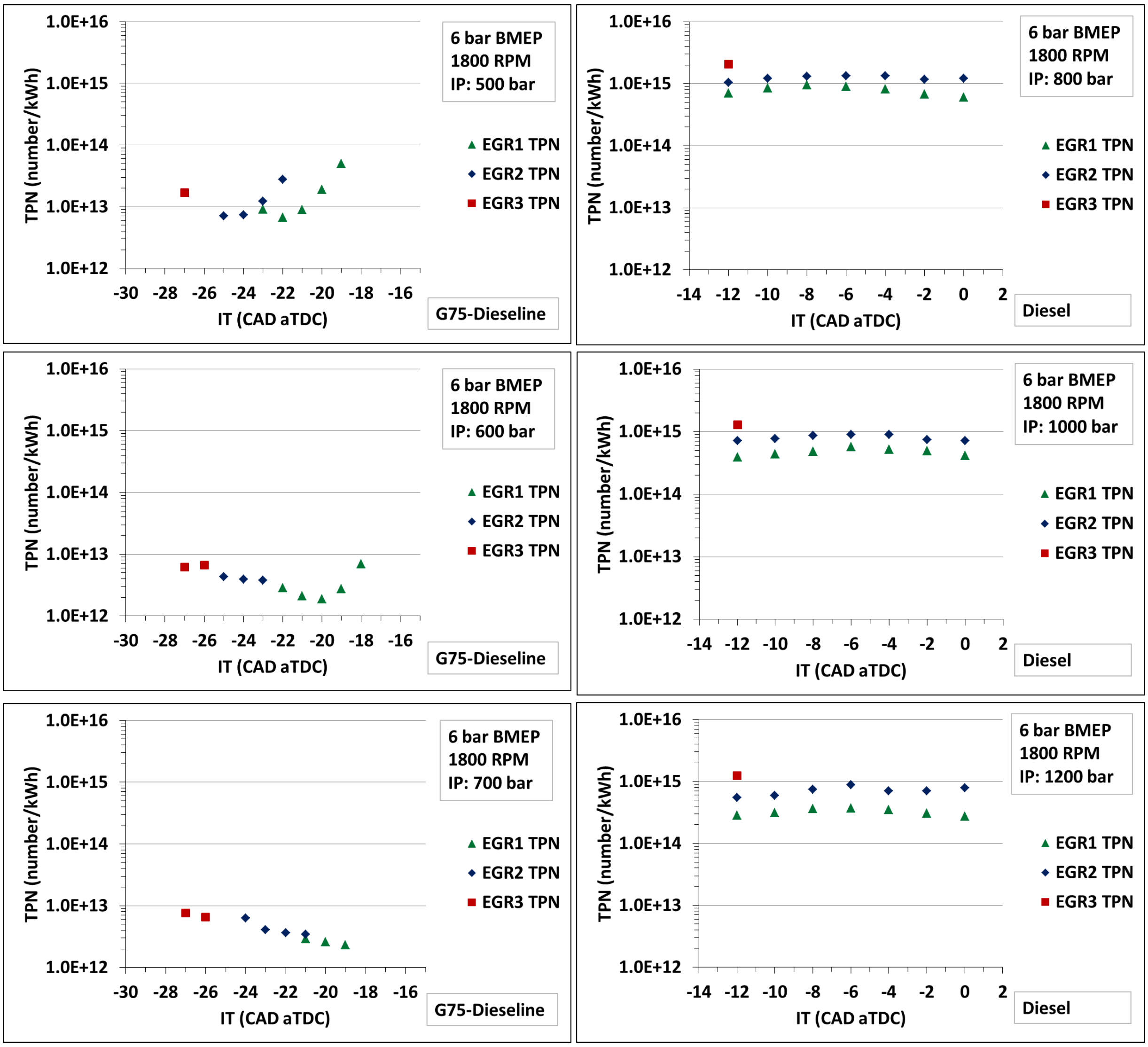



**Figure 6** Brake specific total particle number (TPN) for G75, left column, and diesel, right column, at different injection pressures, injection-timings and EGR valve positions at 6 bar BMEP (it should be noted that this graph is a semi-logarithmic plot)

Figure 7 shows the count median diameter (CMD) of particle emissions from G75 and diesel combustion. Based on these results, TPN values for G75 at the IP of 500 bar were dominated by smaller particles (total CMD is less than 30 nm) mainly in the nucleation mode. Using higher IPs of 600 and 700 bar for G75 resulted in larger total CMDs not because of the increased accumulation mode concentration but decrease in the PN mostly in the nucleation mode. Generally, diesel emissions, compared to G75 emissions, were dominated by larger diameter particles as well as having higher nucleation and accumulation mode concentrations. For diesel, more EGR valve opening resulted in larger particles with CMD up to 83.7 nm indicating the dominance of larger agglomerates resulted from high equivalence-ratio of local fuel-air packets and possible lower rate of soot oxidation.

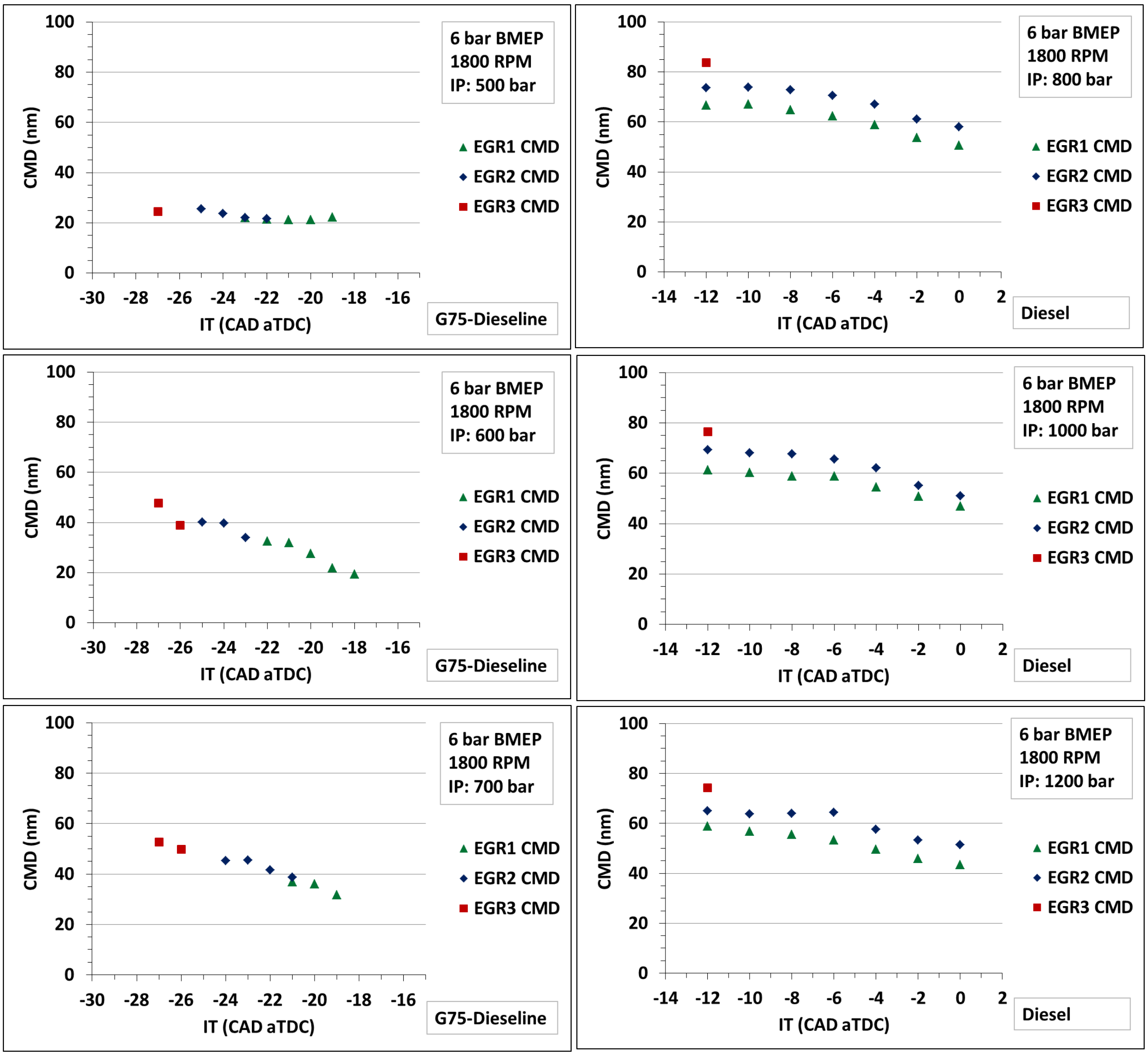
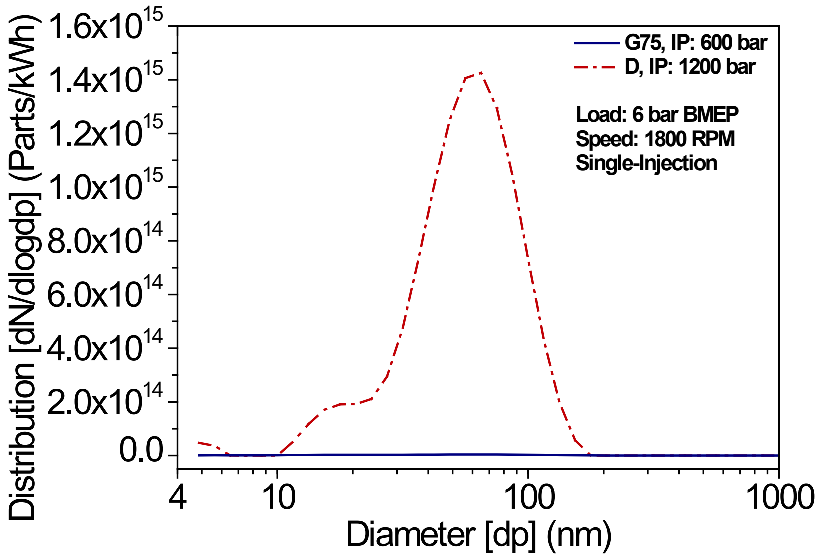



**Figure 7** Particles count median diameter (CMD) for G75, left column, and diesel, right column, at different injection pressures, injection-timings and EGR valve positions at 6 bar BMEP

Figure 8 illustrates the brake specific particle size distribution comparing emissions of G75 and diesel at two selected points.

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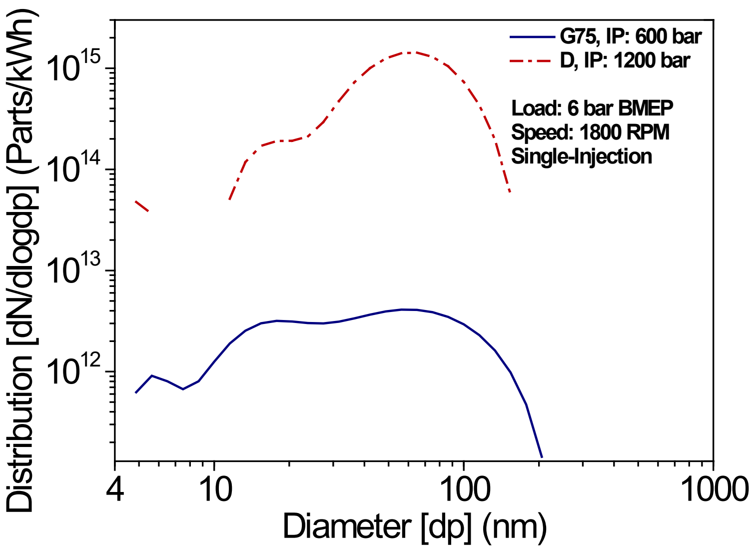


(b)

**Figure 8** Brake specific particle size distributions for G75 and diesel at 6 bar BMEP; (a) semi-logarithmic plot and (b) log-log plot

These points were selected based on achieving minimum smoke and maximum BTE while meeting the target of $BSNO_x$<0.4 g/kWh. Settings for G75 were IP of 600 bar and IT of -24 CAD aTDC and for diesel were IP of 1200 bar and IT of -4 CAD aTDC with the EGR2 valve setting for both. With these settings, diesel and G75 had almost the same BTE and $BSNO_x$ while smoke of diesel was 84 times more than smoke of G75 as shown in Figure 4. Compared to diesel, Acc. BSPM and BSTPN emitted from G75 combustion were approximately 99.5% lower. Based on Figure 8, particles emitted from diesel combustion were mainly in the accumulation mode. No particles were detected in the size range of between approximately 6.5 nm and 10.0 nm. The small sub-6 nm tail in the results can be very small nuclei and their concentration was very low compared to the detected particles in other size bins. Concentrations of particles emitted from G75 combustion were lower than diesel in the entire studied size range. The peaks associated with nucleation and accumulation mode particles were in the same size range when comparing the two fuels, although at much lower concentrations for G75. This can indicate that particle formation process was similar for both fuels but at a much lower rate (especially for accumulation mode particles) and at different regions [3] for low cetane-number fuels.

In general, compared to diesel combustion, G75 combustion resulted in higher BSTHC and BSCO emissions as illustrated in Figure 9 and Figure 10, respectively. In contrast to the case of G75 combustion, variations of BSTHC emissions from diesel combustion were not significant and BSTHC stayed at less than approximately 0.5 g/kWh regardless of IP, EGR valve opening and IT settings. Variations of BSCO emissions for diesel were also insignificant compared to G75, although for both fuels more EGR valve opening increased BSCO slightly. At the selected points discussed earlier, BSTHC and BSCO emissions of G75 was around 4 and 2.3 times higher than diesel, respectively, possibly due to fuel-air overmixing [3].

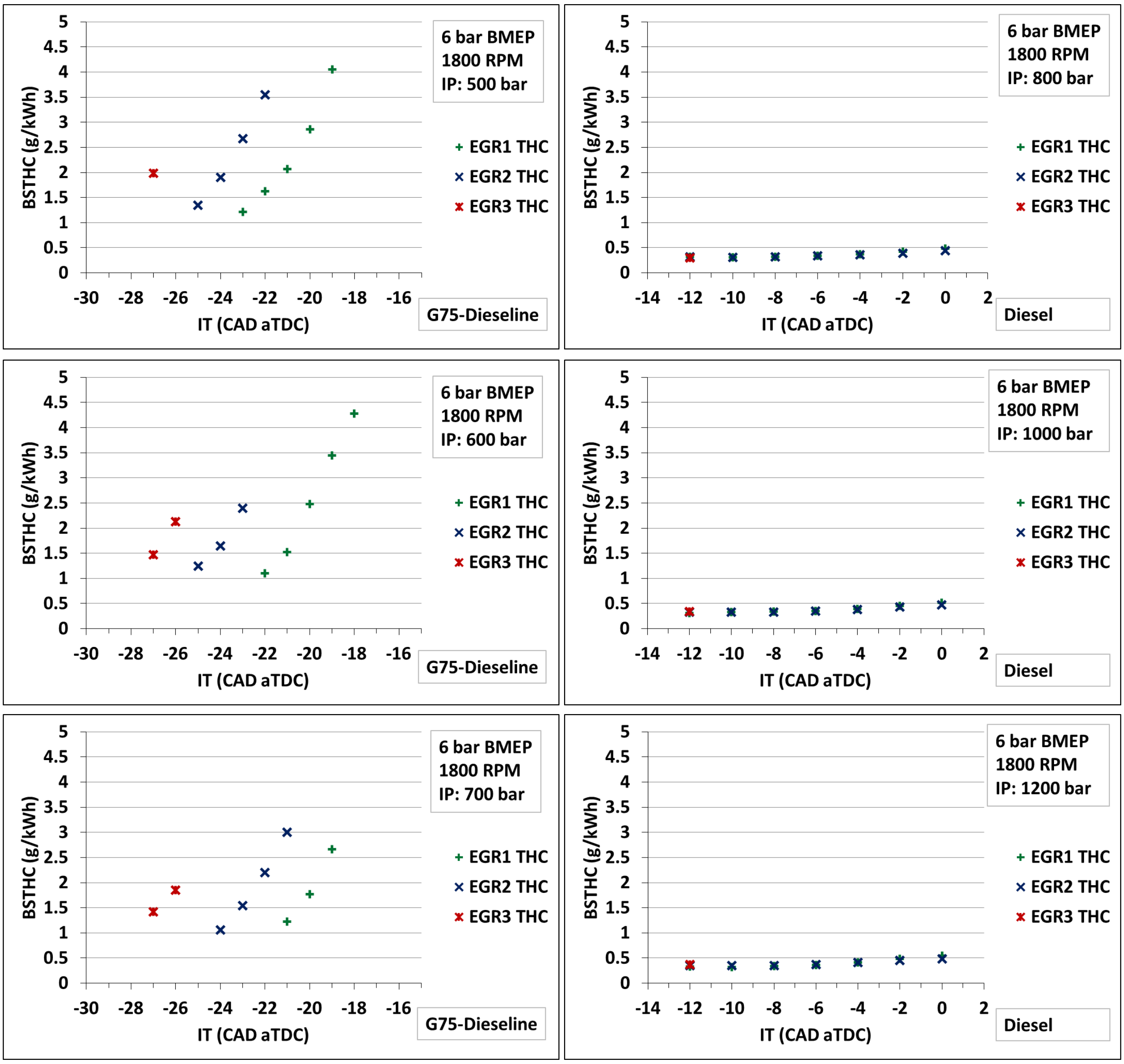


**Figure 9** Brake specific total hydrocarbons (BSTHC) for G75, left column, and diesel, right column, at different injection pressures, injection-timings and EGR valve positions at 6 bar BMEP

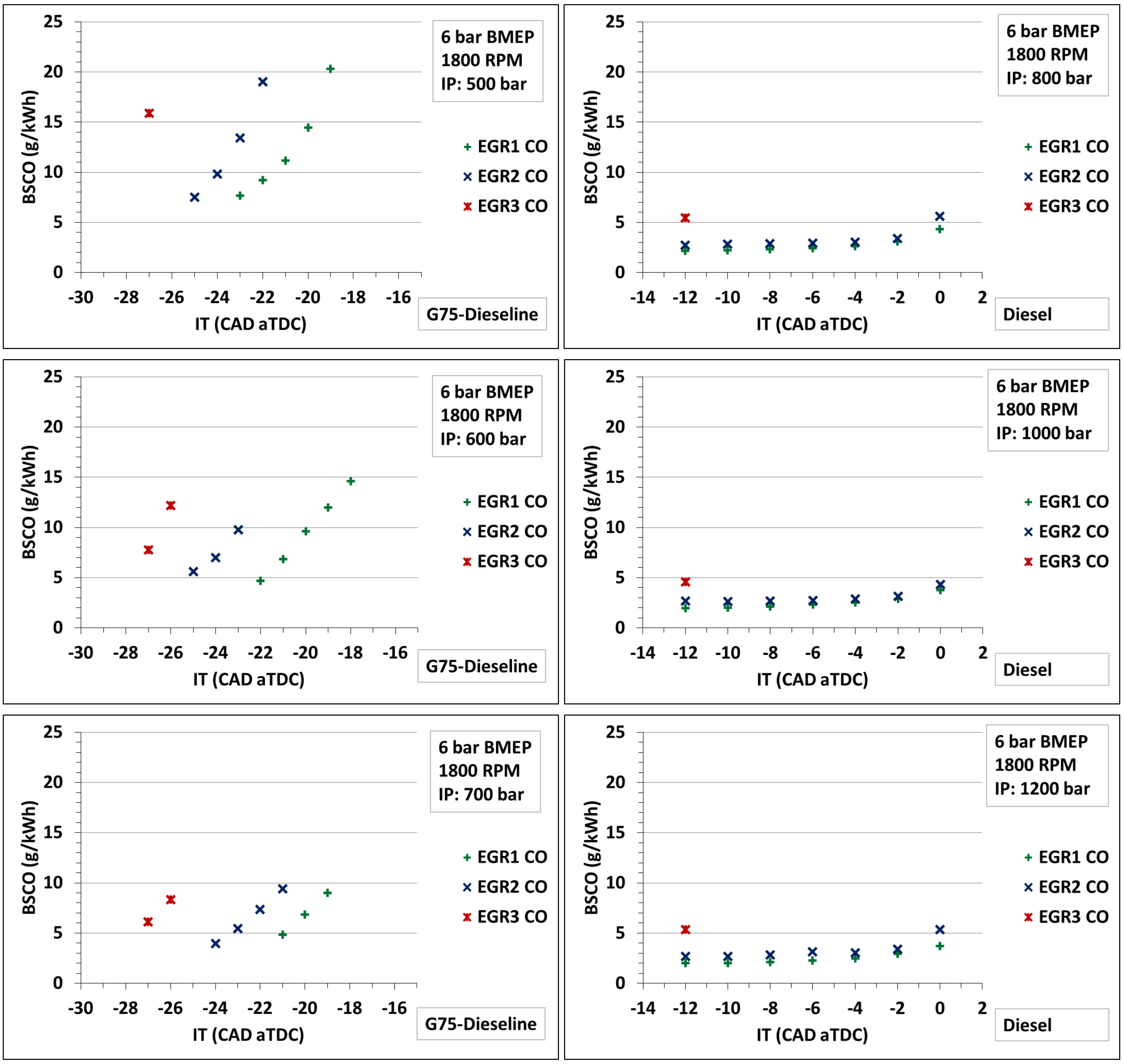


**Figure 10** Brake specific carbon monoxide (BSCO) for G75, left column, and diesel, right column, at different injection pressures, injection-timings and EGR valve positions at 6 bar BMEP

Figure 11 illustrates in-cylinder pressure, heat release rate (HRR) and injection pulse of G75 and diesel at two selected points (similar to what was discussed earlier for minimum smoke and maximum BTE). These points had the same AHR-50 fixed at 12 CAD aTDC. The reason for shorter pulse duration for diesel fuel is the fact that it was injected at a higher pressure compared to G75. It can be seen that the separation between the end of injection (EOI) and the start of combustion (SOC) for G75 was more pronounced compared to diesel owing to its longer ignition-delay at this load. This supports better fuel-air mixing and consequently lower particle emissions. G75 combustion resulted in a higher peak of HRR and in-cylinder pressure compared to diesel due to more premixed combustion. There were some slight differences in the in-cylinder pressure trace of G75 and diesel during the later stages of compression stroke (Figure 11). Different in-cylinder gas temperature history, in-cylinder gas composition and combustion mode for G75 and diesel can be hypothesised to have caused these relatively small differences at this engine load. After the start of G75 injection, there was a reduction of HRR probably due to the cooling effect of fuel. This cooling effect reduced the top dead centre peak pressure compared to diesel which was injected closer to the TDC.

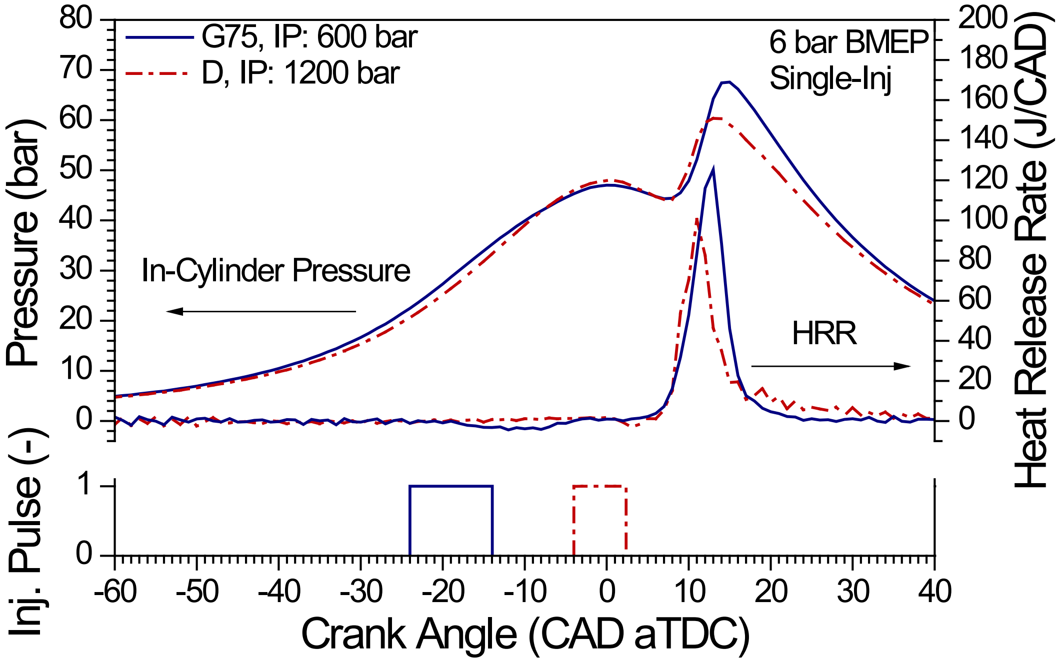


**Figure 11** In-cylinder pressure, heat release rate and injection pulse for G75 and diesel at 6 bar BMEP using single-injection

Based on the results presented for 6 bar BMEP using single-injection, changes in combustion and emissions characteristics of G75 when varying the IT were more significant compared to diesel. This indicates the importance of the AHR-50 for G75 fuel due to its longer ignition-delay. There was a trade-off between THC-CO reduction and $NO_x$ reduction when operating on G75. The main issue for diesel combustion at this load was high smoke emissions which can be reduced at the expense of either increased $BSNO_x$ or reduced BTE. For both fuels there was a trade-off between increasing BTE and decreasing $NO_x$.

## 3.2. Results at 12 bar BMEP

Longer ignition-delay (ID) of G75, compared to diesel, observed at lower loads was less significant at 12 bar BMEP due to higher in-cylinder pressure and temperature. Screening tests at 12 bar BMEP with G75 indicated that there were difficulties with achieving low-$NO_x$ and low-smoke combustion. It was concluded that $\lambda$ had to be always more than 1.2; otherwise smoke emissions were more than 3 FSN and combustion was unstable for G75. Dependence of smoke emissions on this $\lambda$ limit had consequences for $BSNO_x$ emissions. Using high rates of EGR in order to decrease $BSNO_x$, lowered $\lambda$ and consequently higher intake pressure was required to compensate for the reduced intake $O_2$ which was a challenge for the VNT turbocharger. Moreover, producing more intake pressure using the turbocharger means more backpressure for the engine and therefore more pumping losses.

Figure 12 illustrates the results for single-injection of G75 with the IP of 1000 bar at two EGR rates, 15% and 18% (while $\lambda$ was fixed between 1.2 and 1.3), and three different ITs. Using IPs below 1000 bar resulted in high smoke values (more than 3 FSN). In this figure, there is also a base case for diesel single-injection with the same IP for comparison purposes. BTE of G75 combustion at this load was increased by advancing the IT at both rates of EGR. This was due to occurrence of combustion closer to the TDC (advanced AHR-50) which increased BTE as well as MPRR. Smoke emissions followed the same trend in response to advancing the IT and higher EGR rate resulted in elevated smoke emissions as opposed to the observations at 6 bar BMEP. This can be linked with combustion temperature, $O_2$ availability and possible fuel spray-piston interaction. Differences in BSTHC and BSCO emissions from combustion of G75 and diesel were insignificant as shown in Figure 12. These results indicate that G75 emissions were more similar to diesel as opposed to the observations at the lower studied load, although smoke emissions were still lower.

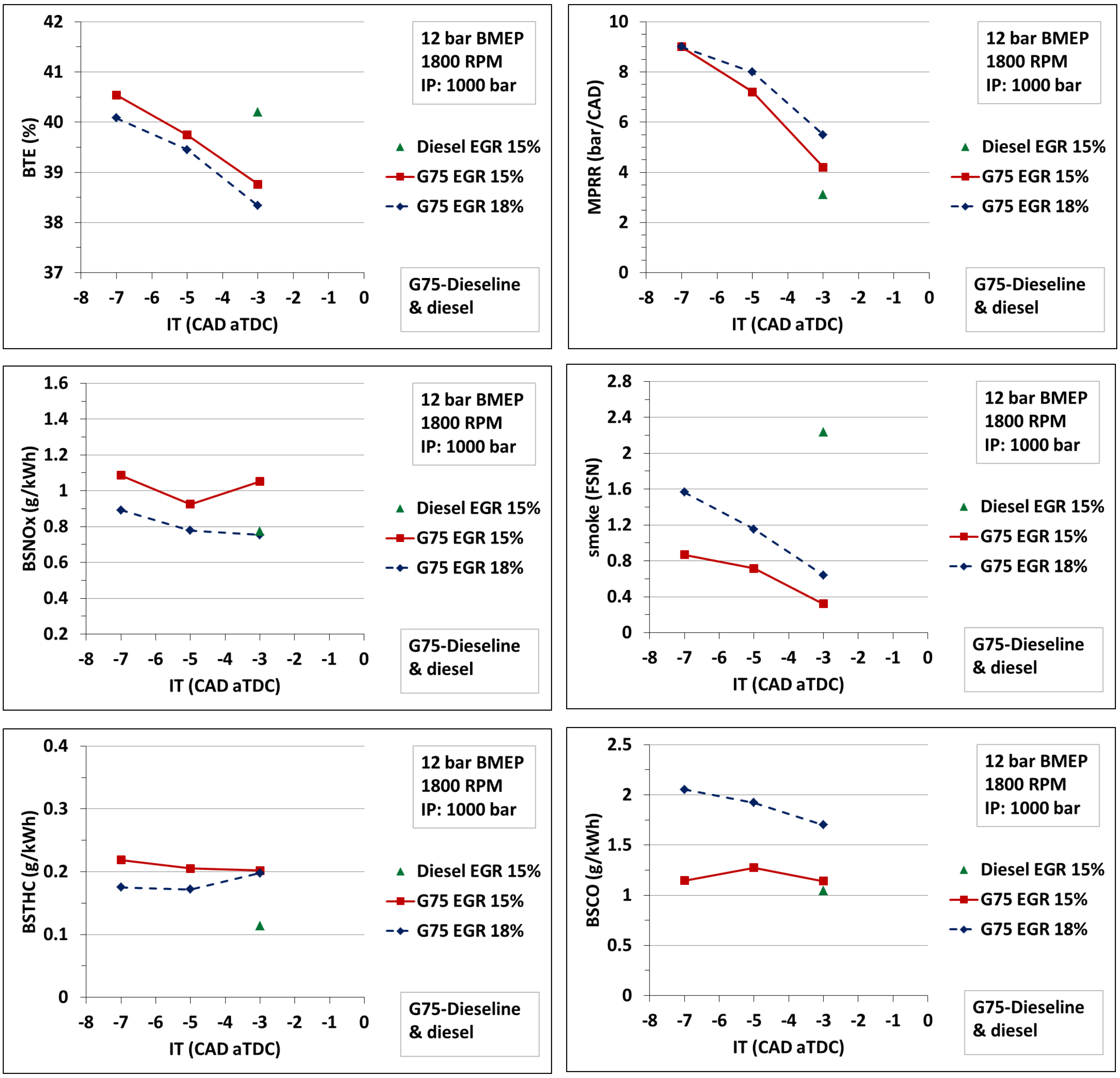


**Figure 12** Comparison of results of G75 with the results of baseline diesel at 12 bar BMEP with fixed injection pressure of 1000 bar

In-cylinder pressure, HRR and injection pulse of G75 were compared to diesel at the same IP, EGR rate and combustion-phasing (Figure 13). Using diesel resulted in mixing-controlled combustion indicated by the negative ignition-dwell and low peak of HRR. G75 combustion was more close to the premixed type combustion with a relatively longer ID, although ignition-dwell was not significantly long. The main outcome of the difference between combustion of these two fuels was the higher smoke emission of diesel shown in Figure 12. From these results, clearly MPRR for G75 combustion was higher than diesel at the same AHR-50, although it was <10 bar/CAD.

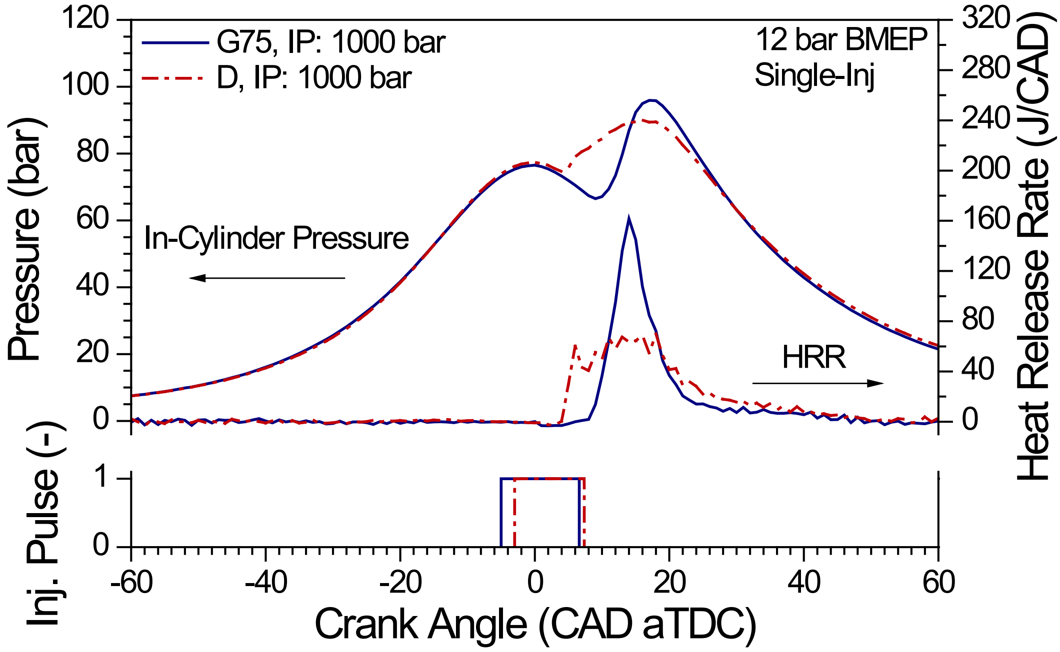


**Figure 13** In-cylinder pressure, heat release rate and injection pulse for G75 and diesel at 12 bar BMEP, IP of 1000 bar and EGR rate of 15%

Increasing IP from 1000 to 1200 bar for both fuels, with fixed λ and AHR-50 at 1.2 and 13 CAD aTDC, respectively, resulted in reduced BTE (Table 3). This can be due to more mechanical load on the engine when using a higher IP. While BTE of both fuels was in the same range, smoke emission of diesel combustion was higher than G75 by around 3.35 times. However, $BSNO_x$ emission of G75 was around 1.6 times more than the results for diesel. This means at this load, there was a trade-off between $NO_x$ and smoke reduction for both of the studied fuels.

Brake specific particle number and size distributions are presented in Figure 14. G75 fuel combustion resulted in less TPN (by around 79.5%) while total CMD was 23.1 nm smaller compared to diesel combustion. Normalised concentrations of particles for G75 (in terms of Parts/kWh) were higher in this load compared to the lower loads. Similar to the results at 6 bar BMEP, bimodal distributions were observed for both fuels while concentrations of particles emitted from G75 combustion were lower than diesel in most of the studied size bins. Accumulation mode concentration peaked at a smaller particle diameter for G75 compared to diesel. It can be hypothesised that this was mainly due to better fuel-air mixing process.

**Table 3** Comparison of G75 and diesel at 12 bar BMEP with fixed injection pressure and combustion-phasing; IT for G75 and diesel were -5 and -3 CAD aTDC, respectively

| Fuel | IP | BTE | $BSNO_x$ | BSTHC | BSCO | smoke |
|---|---|---|---|---|---|---|
| (-) | bar | % | g/kWh | g/kWh | g/kWh | FSN |
| G75 | 1200 | 39.53 | 1.01 | 0.15 | 1.43 | 0.819 |
| Diesel | 1200 | 39.78 | 0.63 | 0.08 | 1.40 | 2.744 |

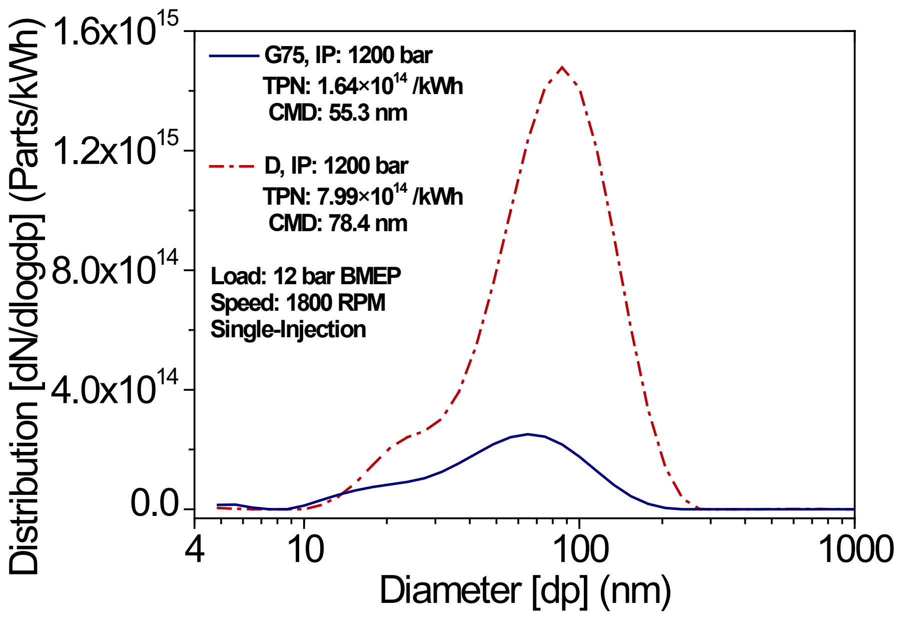


**Figure 14** Brake specific particle number and size distribution for G75 and diesel at 12 bar BMEP

### 3.3. Results at 17.3 bar BMEP

Results at the high load of 17.3 bar BMEP showed similar trends to the results at 12 bar BMEP for G75 fuel combustion (Table 4). When operating on G75 fuel, IPs above 1100 bar could not be achieved. It can be hypothesised that some properties of G75 fuel, most importantly lower viscosity and initial boiling point, could have some effects on the operation of the high pressure fuel pump.

Increase of the EGR rate and retarding the IT reduced BTE and $BSNO_x$ but increased smoke. However, at 12 bar BMEP, smoke was reduced as IT was retarded. This can be explained by considering the possible differences in the rate of soot generation and soot oxidation at these two loads. Earlier ITs resulted in longer IDs, although for the entire studied cases ignition-dwell was negative. It should be mentioned that absolute intake pressure at this load was more than 2.5 bar and therefore ID was shorter [30].

**Table 4** G75 combustion performance and emissions results at 17.3 bar BMEP and IP of 1100 bar

| IT | EGR | BTE | $BSNO_x$ | BSTHC | BSCO | smoke | TPN | Acc. PM |
|---|---|---|---|---|---|---|---|---|
| CAD aTDC | % | % | g/kWh | g/kWh | g/kWh | FSN | Number/kWh | g/kWh |
| -3 | 4.79 | 39.25 | 2.46 | 0.14 | 1.02 | 0.935 | $2.08\times10^{14}$ | 0.068 |
| -3 | 5.46 | 39.04 | 2.11 | 0.09 | 0.96 | 1.024 | $2.02\times10^{14}$ | 0.074 |
| -1 | 5.02 | 37.77 | 1.76 | 0.07 | 1.60 | 2.144 | $3.15\times10^{14}$ | 0.130 |

Results of the G75 combustion were compared to results of the diesel combustion (listed in Table 5) with the same AHR-50 and EGR rate (4.7%). Compared to diesel results, using G75 reduced smoke and TPN emissions by approximately 44.7% and 46.9%, respectively, while $BSNO_x$ emissions were higher by approximately 1.43 times. This is possibly due to more pronounced premixed portion of the G75 combustion as a result of its relatively longer ID compared to diesel. Similar to 12 bar BMEP, increasing the EGR rate was limited by the available intake pressure from the VNT turbocharger. It seems that properties of G75 fuel were not significantly helpful to reduce $NO_x$ and smoke simultaneously at this high load in the current engine. The mixing-controlled nature of the combustion for both fuels resulted in observing the trade-off for $NO_x$ and soot reduction.

**Table 5** Comparison of G75 and diesel at 17.3 bar BMEP with the same AHR-50; IT of G75 and diesel were fixed at -3 and 0 CAD aTDC, respectively

| Fuel | IP | BTE | $BSNO_x$ | BSTHC | BSCO | smoke | TPN | Acc. PM |
|---|---|---|---|---|---|---|---|---|
| (-) | bar | % | g/kWh | g/kWh | g/kWh | FSN | Number/kWh | g/kWh |
| G75 | 1100 | 39.25 | 2.46 | 0.14 | 1.02 | 0.935 | $2.08\times10^{14}$ | 0.068 |
| Diesel | 1200 | 38.96 | 1.72 | 0.06 | 0.95 | 1.692 | $3.92\times10^{14}$ | 0.146 |

Figure 15 illustrates in-cylinder pressure, HRR and injection pulse for both fuels at 17.3 bar BMEP with settings used in Table 5. SOC for both fuels was before the EOI resulting in a lower peak of HRR compared to 12 bar BMEP as less premixing was achieved.

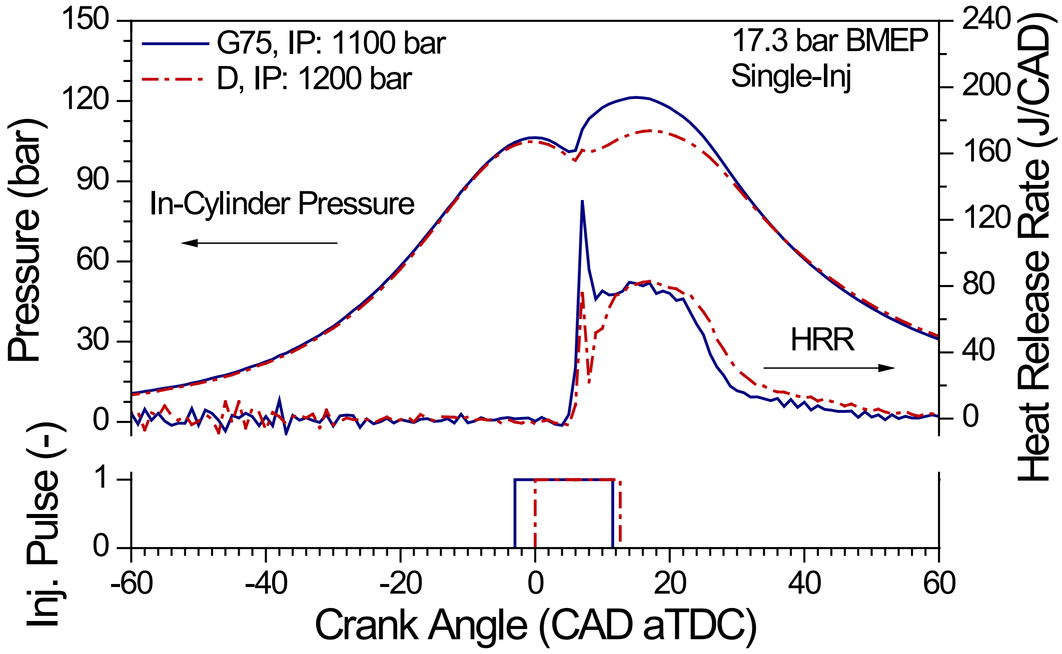


**Figure 15** In-cylinder pressure, heat release rate and injection pulse for G75 and diesel at 17.3 bar BMEP

It can be concluded that G75 and diesel fuels illustrated similar combustion and emissions characteristics at high loads. It is suggested that, for high loads, higher intake pressure should be investigated while considering effects of the imposed higher backpressure on engine efficiency. It is expected that with higher intake pressure, more EGR can be used and therefore $NO_x$ will be reduced while intake $O_2$ concentration is high enough to either reduce soot formation or improve soot oxidation [3, 28, 31, 32]. Moreover, interactions between the fuel spray and piston and/or cylinder walls need to be studied for G75. Hole diameter and included angle of the injector as well as the geometry of piston and cylinder head are required to be optimised to suit G75 fuel and its combustion. Optimisation of the fuel injection and EGR strategies for G75 combustion can be another area of development. Furthermore, since combustion of G75 and diesel is more similar at high loads, there is a scope for studying fuel composition effects, e.g HC structure, aromatics content and oxygenates content, on particle emissions [31, 33]. Addition of viscosity improver to the G75 can be investigated to identify any improvements on achieving higher injection pressures at high loads.

# 4 SUMMARY AND CONCLUSIONS

Compression ignition (CI) combustion of G75-Dieseline (G75) and diesel in a light-duty 4-cylinder engine has been investigated at a fixed engine speed of 1800 RPM and at 6, 12 and 17.3 bar BMEP. The major findings are as follows:

- Particle emissions from G75 combustion were lower than diesel combustion by up to 99.5% in both number and mass, while BTE and $NO_x$ remained in the same range. This was mainly due to the longer ignition-delay and higher volatility of G75.
- Bimodal particle size distributions (nucleation and accumulation modes) were observed for both fuels while particle concentrations (especially accumulation mode) for G75 were much lower at the entire particle size range.
- The reduction of particle number emissions caused by increasing the fuel injection pressure was less evident in the accumulation mode compared to the nucleation mode for G75. The variation trend of particle mass emissions was similar to smoke.
- At medium loads, premixed combustion and emissions of G75 were more sensitive to the fuel injection timing compared to diesel due to its longer ignition-delay and ignition-dwell.

- At high loads (especially 17.3 bar BMEP in this study), the mixing-controlled combustion following the phase of premixed combustion was observed for G75, although less pronounced than in diesel combustion, because of shorter ignition-delay compared to lower loads.

There is a scope for optimising the intake pressure boosting system, piston/cylinder-head geometries, injector included angle and fuel injection strategies for G75 combustion. These will be helpful for further reduction of $NO_x$ and smoke emissions at high operating loads.

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# APPENDIX

## Nomenclature

| | |
|---|---|
| Acc. PM | Accumulation mode particle mass |
| AHR-50 | Combustion-phasing (defined as the CAD at which 50% of the accumulative heat release is achieved) |
| aTDC | After top dead centre |
| BMEP | Brake mean effective pressure |
| BS | Brake specific |
| BTE | Brake thermal efficiency |
| CAD | Crank angle degree |
| CI | Compression ignition |
| CMD | Count median diameter |
| CN | Cetane-number |
| CO | Carbon monoxide |
| $CO_2$ | Carbon dioxide |
| DAQ | Data acquisition board |
| DI | Direct injection |
| Dieseline | A blend of diesel and gasoline |
| ECU | Engine control unit |
| EGR | Exhaust gas recirculation |
| EOI | End of injection |
| FSN | Filter smoke number |
| G75 | A blend of 75% gasoline in diesel based on volume |
| HC | Hydrocarbons |

| | |
|---|---|
| HRR | Heat release rate |
| ID | Ignition-delay |
| IP | Injection pressure |
| IT | Injection-timing |
| MPRR | Maximum pressure rise rate |
| $NO_x$ | Oxides of nitrogen |
| $O_2$ | Oxygen |
| PCI | Premixed compression ignition |
| RON | Research octane number |
| RPM | Revolutions per minute |
| SD | Standard deviation |
| SOC | Start of combustion |
| TDC | Top dead centre |
| THC | Total hydrocarbons |
| TPN | Total particle number |
| VNT | Variable-nozzle-turbine |
| $\lambda$ | Specific air-fuel ratio (actual air/fuel ratio over stoichiometric air/fuel ratio) |